

A four power-piston low-temperature differential Stirling engine using simulated solar energy as a heat source

Bancha Kongtragool, Somchai Wongwises*

Fluid Mechanics, Thermal Engineering and Multiphase Flow Research Laboratory (FUTURE), Department of Mechanical Engineering, Faculty of Engineering, King Mongkut's University of Technology Thonburi, Bangmod, Bangkok 10140, Thailand

Received 21 August 2007; received in revised form 3 December 2007; accepted 14 December 2007
Available online 11 January 2008

Communicated by: Associate Editor Robert Pitz-Paal

Abstract

In this paper, the performances of a four power-piston, gamma-configuration, low-temperature differential Stirling engine are presented. The engine is tested with air at atmospheric pressure by using a solar simulator with four different solar intensities as a heat source. Variations in engine torque, shaft power and brake thermal efficiency with engine speed and engine performance at various heat inputs are presented. The Beale number obtained from the testing of the engine is also investigated. The results indicate that at the maximum actual energy input of 1378 W and a heater temperature of 439 K, the engine approximately produces a maximum torque of 2.91 N m, a maximum shaft power of 6.1 W, and a maximum brake thermal efficiency of 0.44% at 20 rpm.

© 2008 Published by Elsevier Ltd.

Keywords: Stirling engine; Hot-air engine; Solar-powered heat engine; Solar simulator

1. Introduction

The low-temperature differential (LTD) Stirling engine is a type of Stirling engine that can run with a small temperature difference between the hot and cold ends of the displacer cylinder. The LTD Stirling engine is therefore able to operate with various low-temperature heat sources.

Some characteristics of the LTD Stirling engine are as follows:

- (1) Displacer to power-piston swept volumes ratio or compression ratio is large.
- (2) Diameters of displacer cylinder and displacer are large.
- (3) Displacer length is short.
- (4) Effective heat transfer surfaces on both end plates of the displacer cylinder are large.

- (5) Displacer stroke is small.
- (6) Dwell period at the end of the displacer stroke is slightly longer than the normal Stirling engine.
- (7) Operating speed is low.

While the Stirling engine has been studied by a large number of researchers, the LTD Stirling engine has received comparatively little attention. Many studies related to solar-powered Stirling engines and LTD Stirling engines have been reviewed in the authors' previous works (Kongtragool and Wongwises, 2003a). Some of these works are described as follows:

Haneman (1975) studied the possibility of using air with low-temperature sources. This led to the construction of an unusual engine, in which the exhaust heat was still sufficiently hot to be useful for other purposes.

A simply constructed low-temperature heat engine modeled on the Stirling engine configurations was patented by White (1983). White suggested improving performance by pressurizing the displacer chamber. Efficiencies were

* Corresponding author. Tel.: +662 4709115; fax: +662 4709111.
E-mail address: somchai.won@kmutt.ac.th (S. Wongwises).

Nomenclature

A	absorber area (m^2)	q	total heat input from heat source (W)
c_p	specific heat of water at constant pressure (4186 J/kg K)	r	dynamometer brake drum radius (m)
E_H	heat source efficiency	S	spring balance reading (N)
E_{BT}	brake thermal efficiency	T	engine torque (N m)
f	engine frequency (Hz)	T_C	cooler wall temperature (K)
I	average intensity on absorber plate (W/m^2)	T_H	heater wall temperature (K)
m_w	mass of water to absorb heat (kg)	T_{w1}	initial water temperature (K)
N	engine speed (rpm, rps)	T_{w2}	final water temperature (K)
N_B	Beale number ($\text{W}/\text{bar cm}^3 \text{ Hz}$)	t_1	initial time at water temperature of T_{w1} (s)
P	shaft power (W)	t_2	final time at water temperature of T_{w2} (s)
p_m	engine mean-pressure (bar)	V_P	power-piston swept volume (cc)
q_{in}	actual heat input to the engine (W)	W	loading weight (N)

claimed to be around 30%, which is regarded as quiet high for a low-temperature engine.

O'Hare (1984) patented a device which passed cooled and heated streams of air through a heat exchanger by changing the pressure of air inside the bellows. The practical usefulness of this device was not shown in detail as in the case of Haneman's work. Spencer (1989) reported that, in practice, such an engine would produce only a small amount of useful work relative to the collector system size, and would give little gain compared to the additional maintenance required.

Senft's work (Senft, 1991) showed the motivation in the use of Stirling engine. Their target was to develop an engine operating with a temperature difference of 2 °C or lower. Senft (1993) described the design and testing of a small LTD Ringbom Stirling engine powered by a 60° conical reflector. He reported that the tested 60° conical reflector, producing a hot end temperature of 93 °C under running conditions, worked very well.

Rizzo (1997) reported that Kolin experimented with 16 LTD Stirling engines, over a period of 12 years. Kolin presented a model that worked on a temperature difference between the hot and cold ends of the displacer cylinder which was as low as 15 °C. Iwamoto et al. (1997) compared the performance of a LTD Stirling engine with a high-temperature differential Stirling engine. They concluded that the LTD Stirling engine efficiency at its rated speed was approximately 50% of the Carnot efficiency. However, the compression ratio of their LTD Stirling engine was approximately equal to that of a conventional Stirling engine. Its performance, therefore, seemed to be the performance of a common Stirling engine operating at a low operating temperature.

Senft Van Arsdell (2001) made an in-depth study of the Ringbom engine and its derivatives, including the LTD engine. Senft's research into LTD Stirling engines resulted in an interesting engine, which had an ultra-low temperature difference of 0.5 °C. It has been very difficult for anyone to create an engine with a result better than this.

Kongtragool and Wongwises (2003b) investigated the Beale number for LTD Stirling engines by collecting the existing Beale number data for various engine specifications from the literature. They concluded that the Beale number for a LTD Stirling engine could be found from the mean-pressure power formula.

Kongtragool and Wongwises (2005a) theoretically investigated the power output of a gamma-configuration LTD Stirling engine. Former works on Stirling engine power output calculations were studied and discussed. They pointed out that the mean-pressure power formula was the most appropriate for LTD Stirling engine power output estimation. However, the hot-space and cold-space working fluid temperatures were needed in the mean-pressure power formula.

Kongtragool and Wongwises (2005b) presented the optimum absorber temperature of a once-reflecting full-conical reflector for a LTD Stirling engine. A mathematical model for the overall efficiency of a solar-powered Stirling engine was developed and the limiting conditions of both maximum possible engine efficiency and power output were studied. Results showed that the optimum absorber temperatures obtained from both conditions were not significantly different. Furthermore, the overall efficiency in the case of the maximum possible engine power output was very close to that of the real engine of 55% Carnot efficiency.

Kongtragool and Wongwises (2007a) also reported the performance of two LTD Stirling engines tested using LPG gas burners as heat sources. The first engine was a twin-power-piston engine and the second one was a four-power-piston engine. Engine performances, thermal performances, including the Beale's numbers were presented.

Recently, Kongtragool and Wongwises (2007b) presented the performance of a twin-power-piston Stirling engine powered by a solar simulator. This engine was the same as the engine described in (Kongtragool and Wongwises, 2007a). However, the heat source was a solar simulator made from a 1000 W halogen lamp. Comparisons were made between the characteristics of the

high-temperature differential (HTD) and LTD Stirling engine and methods for performance improvement were also discussed.

Although some information is currently available on the LTD Stirling engine, there still remains room for further research. In particular a detailed investigation is lacking into the LTD Stirling engine using solar energy as a heat source. As a consequence, in this paper, the testing of the performance of a LTD Stirling engine using simulated solar energy is presented. The LTD Stirling engine tested in this paper is a kinematics, single-acting, four power-piston, gamma-configuration. Non-pressurized air is used as a working fluid and a solar simulator fabricated from four 1000 W tungsten halogen lamps is used as a heat source. Since the gamma-configuration provides a large regenerator heat transfer area and is easy to be constructed, this is configuration which is used in this study.

2. Experimental apparatus and procedure

The engine schematic diagram and main design parameters are shown in Fig. 1 and Table 1, respectively. To eliminate the machining difficulties experienced with a single large power-piston, it is designed with four single-acting power-pistons. Two power-pistons are connected with piston rods and a flat bar (see Fig. 2). Four power cylinders are directly connected to the cooler plate to minimize the cold-space and dead volume transfer-port. Furthermore, the cooler plate is a part of the cooling water pan.

In order to make the engine compact and to minimize the number of engine parts, a simple crank mechanism is used in this engine. The crankshaft, which is supported by two ball bearings, is made from a steel shaft, two crank discs and a crank pin. The crank pin is connected to the displacer connecting rod. Two steel flywheels, which also

Table 1
Engine main design parameters

Mechanical configuration	Gamma
Power piston	
bore (cm) × stroke (cm)	13.3 × 13.3
swept volume (cm ³)	7391
Displacer	
bore (cm) × stroke (cm)	60 × 14.48
swept volume (cm ³)	40,941
Compression ratio	5.54
Phase angle	90°

act as the crank discs for the power-pistons, are attached to both ends of the crankshaft.

The power cylinders and pistons are made from steel. The piston surfaces have brass lining and oil grooves, 1 mm × 1 mm with 10 mm spacing. The clearance between piston and bore is approximately 0.02 mm. The displacer cylinder and head is made from a 1 mm thick stainless steel plate and the clearance between them is 2 mm. The displacer also serves as a regenerator, which is made from a round-hole perforated steel sheet. The stainless steel pot scourer is used as a regenerator matrix.

The displacer rod, made from a stainless steel pipe, is guided by two brass bushings placed inside the displacer rod guide house. Leakage through these bushings is prevented by two rubber seals. Both ends of the power-piston and displacer connecting rod which are made from steel, are fitted with two ball bearings. Details of the testing facilities are shown in Fig. 2. The intensity placed on the absorber plate (or displacer head) is measured by a pyranometer (Lambert model 00.16103.000000 CM3, calibrated constant of which is 23.66 $\mu\text{V}/\text{Wm}^{-2}$). The sensitivity of the intensity measurement obtained from the pyranometer is $\pm 0.05\%$.

The cooler temperature (T_C) and heater temperature (T_H) are measured by T-type and K-type thermocouples, respectively. The accuracy of temperature measurement is $\pm 0.1^\circ\text{C}$. Four 1000 W tungsten halogen lamps (Osram Haloline 64740 L J R7s) are used as a solar simulator. A data logger (DataTaker model DT 50) is used to collect data from thermocouples and pyranometer.

The engine torque is measured by a rope-brake dynamometer. A displacer crank disc, which is 8.95 cm in radius, is used as a brake drum. The braking load is measured by the loading weight and spring balance reading. A photo tachometer with ± 0.1 rpm accuracy is used to measure the engine speed. The engine tests are performed using four distances from the lamp to the absorber. The average simulated intensities (I) on the absorber plate are 5380, 5772, 6495, and 7094 W/m^2 . The actual heat input to the engine (q_{in}), at the above mentioned intensities, is experimentally determined by using water to absorb this heat. The concentrated heat (q) on the absorber plate, actual heat input into the engine (q_{in}), absorber temperature, and the engine performance (P_{max}) resulting from these simulated intensities are shown in Table 2.

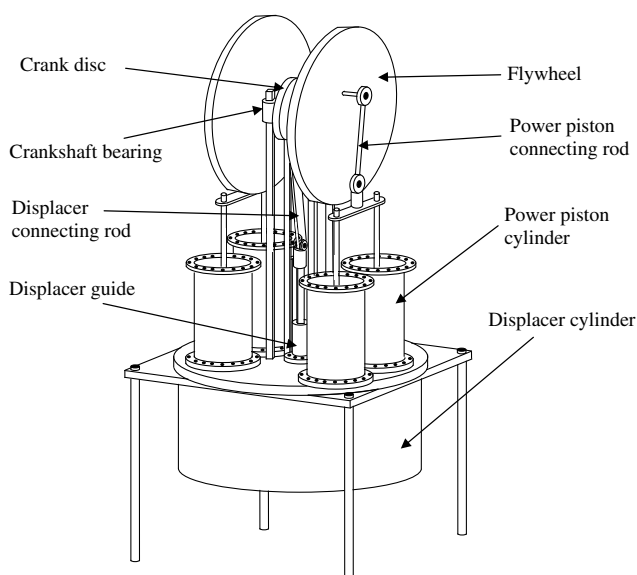


Fig. 1. Schematic diagram of the tested Stirling engine.

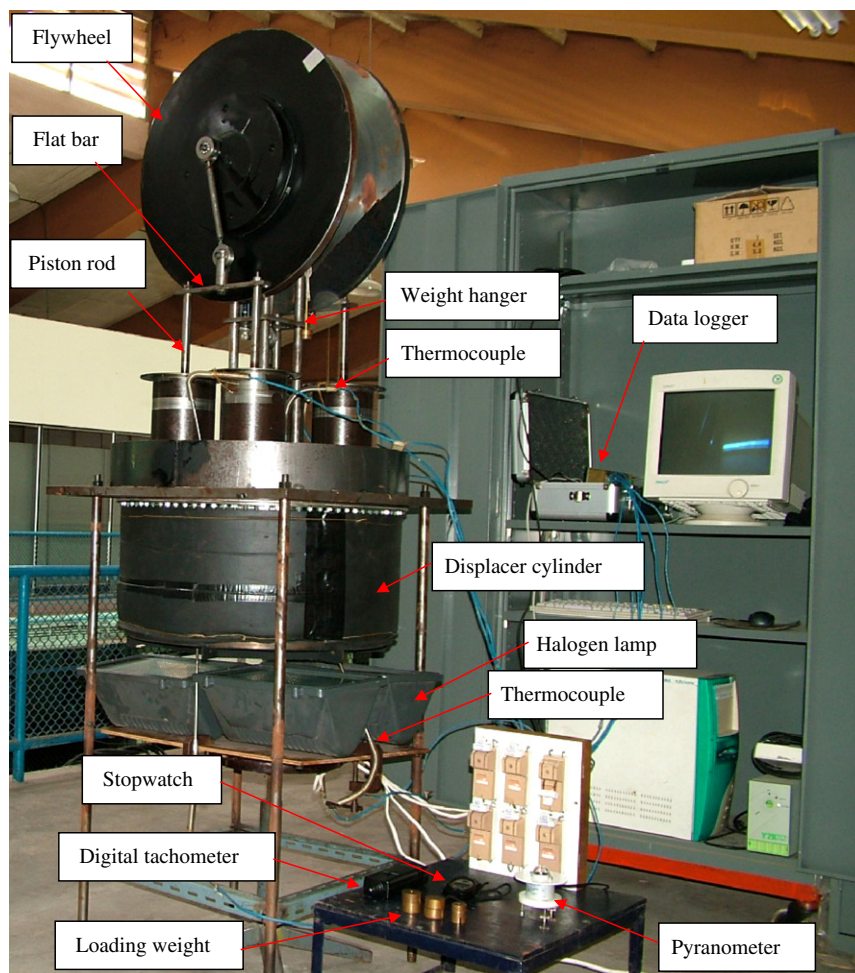


Fig. 2. Engine with testing facilities.

Table 2

Maximum engine performance and Beale number at $T_C = 307$ K

I (W/m ²)	q (W)	q_{in} (W)	T_H (K)	T_{max} (N m)	P_{max} (W)	E_{BTmax} (%)	N_B (W/bar cm ³ Hz)
5380	1521	1235	401	2.21 at 19.0 rpm	4.39 at 19.0 rpm	0.36 at 19.0 rpm	1.8757×10^{-3}
5772	1632	1272	412	2.96 at 15.3 rpm	4.87 at 18.8 rpm	0.38 at 18.8 rpm	2.1029×10^{-3}
6495	1837	1323	425	2.78 at 18.5 rpm	5.44 at 19.6 rpm	0.41 at 19.6 rpm	2.2532×10^{-3}
7094	2006	1378	439	2.91 at 20.0 rpm	6.10 at 20.0 rpm	0.44 at 20.0 rpm	2.4760×10^{-3}

3. Experimental procedures

3.1. Intensity test

A measurement of the actual intensity placed on the absorber plate is needed for the engine performance calculation. The experiment for determination of the actual intensity on the engine absorber at various distances from halogen lamp to absorber was carried out first. A pyranometer was used to measure the intensity on the displacer cylinder head that acted as the absorber plate. A data logger and a personal computer were used to collect data from the pyranometer. The schematic diagram of this test is shown in Fig. 3. The testing procedure was as follows:

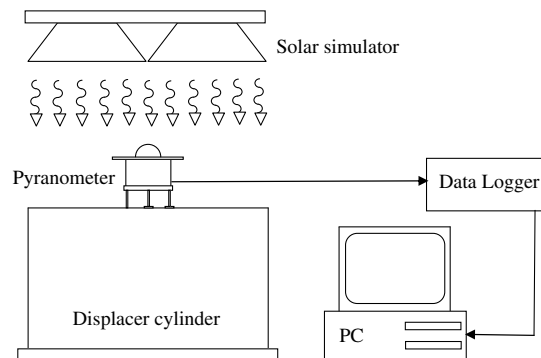


Fig. 3. Schematic diagram of the simulated solar intensity test.

- The displacer cylinder and the halogen lamp were put on the stand.
- The distance from the halogen lamp to the absorber plate was set as required.
- The pyranometer was placed on the absorber plate at 17 positions as shown in Fig. 4.
- The pyranometer was connected to the data logger and computer.
- The halogen lamp was turned on and the intensity was collected at that position.
- The pyranometer was placed to other positions.
- The testing was repeated for other intensities by changing the distance between lamp and absorber.

The test results from those twelve distances are shown in Fig. 7.

3.2. Heat source test

The actual or useful heat input can not be determined directly while the engine is running due to difficulties caused by instrumentation. In order to determine the actual heat input to the engine, therefore, this experiment was carried out before the real performance test had begun. The schematic diagram of the heat source test is shown in Fig. 5. The testing procedure was as follows:

The displacer cylinder, insulated with 25.4 mm thick insulation was put on the stand. Thirty kg of water was poured into the displacer cylinder. This water was used to absorb the heat from the solar simulator. The absorbed heat was the useful heat input to the displacer cylinder.

The thermocouples for measuring the displacer cylinder head wall temperature and the water temperature were installed. Three T-type thermocouples were used to measure the water temperature in the displacer cylinder, while three K-type thermocouples were used to measure the displacer cylinder head wall temperatures.

The halogen lamp was placed at the required distance, underneath the displacer cylinder head. The initial water temperature was recorded and the halogen lamp was turned on. Before the boiling point was reached, all temperatures were taken at every 1-min interval using a data logger and personal computer.

The testing was repeated with another heat input by changing the distance between the halogen lamp and the

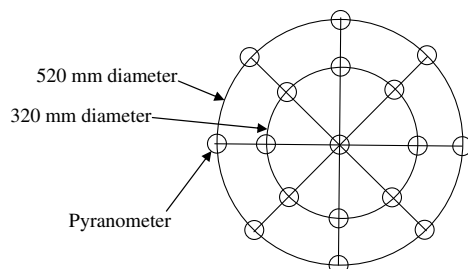


Fig. 4. Positions of pyranometer in the simulated solar intensity test.

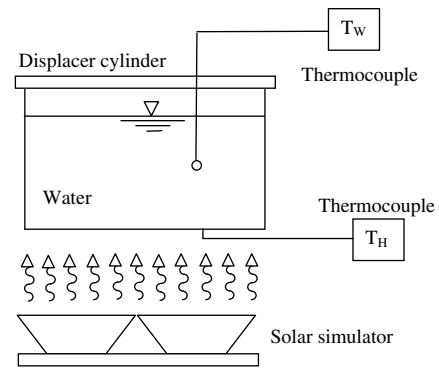


Fig. 5. Schematic diagram of the heat source efficiency test.

absorber. The test results from four intensities are shown in Table 2.

The heat source efficiency (E_H) can be determined from the following equation (Kongtragool and Wongwises, 2007b):

$$E_H = \frac{q_{in}}{q} = \frac{m_w c_p (T_{w2} - T_{w1})}{IA(t_2 - t_1)} \quad (1)$$

where m_w is the mass of water to absorb heat transferred from the heat source, c_p is the specific heat of water at constant pressure, T_{w1} and T_{w2} are the initial water temperature and the maximum water temperature, respectively, t_1 and t_2 are initial and final times at the water temperature of T_{w1} and T_{w2} , respectively, I is the intensity from a solar simulator, and A is the absorber area.

3.3. Performance test

The schematic diagram of the engine performance test is shown in Fig. 6. Before the engine was started, all thermocouples were connected to the data logger and computer

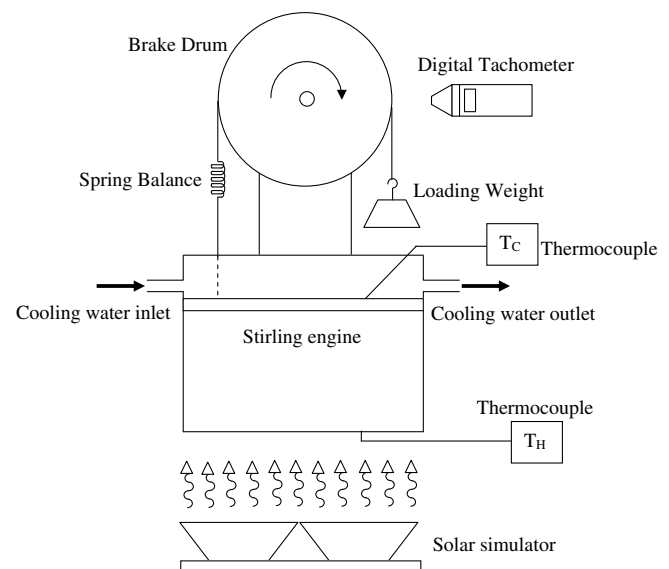


Fig. 6. Schematic diagram of the four power-piston Stirling engine performance test by a solar simulator.

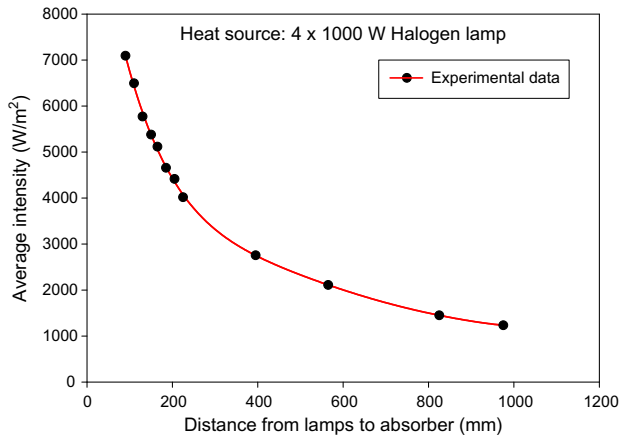


Fig. 7. Average intensity on absorber plate versus distance from lamp to absorber.

and the cooling water system was connected to the engine cooling pan. The cooling water flow rate was adjusted in order to keep water level in the cooling pan constant. Some lubricating oil was ejected into the power-pistons, cylinders, and the displacer guide bushing.

The solar simulator was placed underneath the displacer head at a specified distance. The halogen lamp was then switched on. The displacer head was heated up until it reached the operating temperature. The engine was then started and run until a steady condition was reached.

The engine was loaded by adding a weight to the dynamometer. After that, the engine speed reading, spring balance reading and all temperatures reading from the thermocouples were collected. Another loading weight was added to the dynamometer until the engine was stopped. The actual shaft power (P) can be calculated from:

$$P = 2\pi TN = 2\pi(S - W)rN \quad (2)$$

where T is the engine torque, S is the spring balance reading, W is the loading weight, r is the brake drum radius, and N is the engine speed.

Testing was then repeated with another simulated intensity by changing the distance from the lamp to absorber.

The actual heat input to the engine (q_{in}) at the above mentioned intensities, was experimentally determined by using water to absorb the heat. The concentrated heat (q) on absorber plate, actual heat input to the engine, absorber temperature, and engine performance resulting from these simulated intensities are shown in Table 2. In this table, the brake thermal efficiency E_{BT} is calculated from:

$$E_{BT} = P/q_{in} \quad (3)$$

The Beale number is calculated from the Beale formula (Kongtragool and Wongwises, 2003b, 2005a):

$$N_B = P/(p_m V_P f) \quad (4)$$

Where p_m is engine mean-pressure, V_P is power-piston swept volume and f is engine frequency. For non-pressur-

ized engines, the $p_m = 1$ bar is used in the calculation as described by Senft (1993).

4. Experimental results and discussion

In the engine test, as the load is gradually applied to the engine, its speed is gradually reduced, until eventually it stops. The characteristics are shown in the form of the variation of torque, shaft power and brake thermal efficiency with the engine speed. Only engine performance at the maximum average simulated intensity is presented as a typical performance as shown in Fig. 8.

From Fig. 8, it can be noted that the engine torque decreases with increasing engine speed. Furthermore, the shaft power increases with increasing engine speed until the maximum shaft power is reached and then decreases with increasing engine speed. This decreasing shaft power after the maximum point, results from the friction that increases with increasing speed together with inadequate heat transfer at higher speed. Since the brake thermal efficiency is the shaft power divided by a constant heat input, the curve of brake thermal efficiency has the same trend as the shaft power.

Figs. 9–11 show the variations of engine torque, shaft power and brake thermal efficiency with engine speed at various heat inputs, respectively. As expected, greater engine performance results from the higher heat input. An increase of the engine torque, shaft power and brake thermal efficiency is shown to also depend on the heater temperature.

In Fig. 12, the maximum shaft power and Beale number at various heat inputs are plotted against the heater temperature. As shown in this figure, the shaft power and Beale number increase with an increase in heater temperature.

Results from this study indicate that the engine performance and heater temperature increase with increasing simulated solar intensity. In fact, it can be said that the

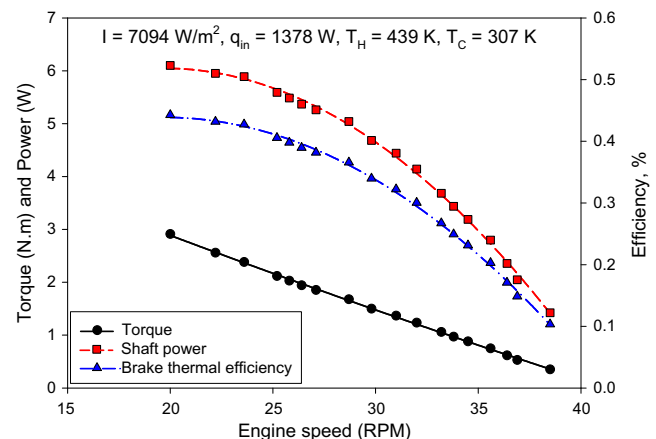


Fig. 8. Engine performance at 7094 W/m² average intensity, 1378 W actual heat input.

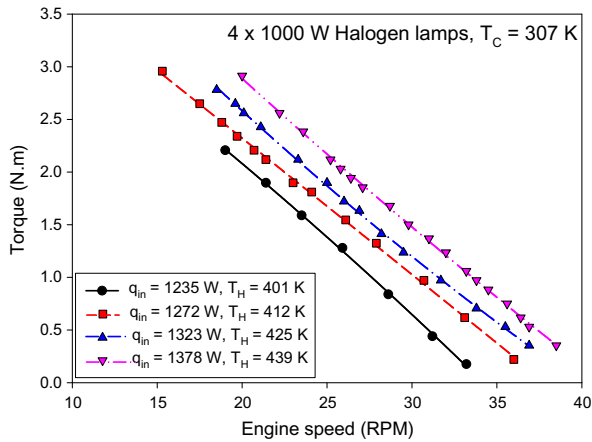


Fig. 9. Variations in engine torque at various actual heat inputs.

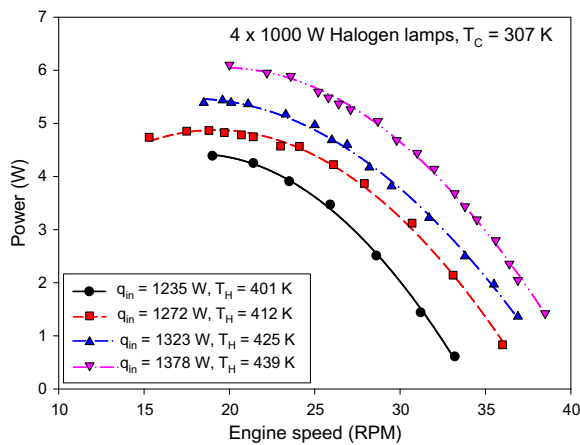


Fig. 10. Variations in engine shaft power at various actual heat inputs.

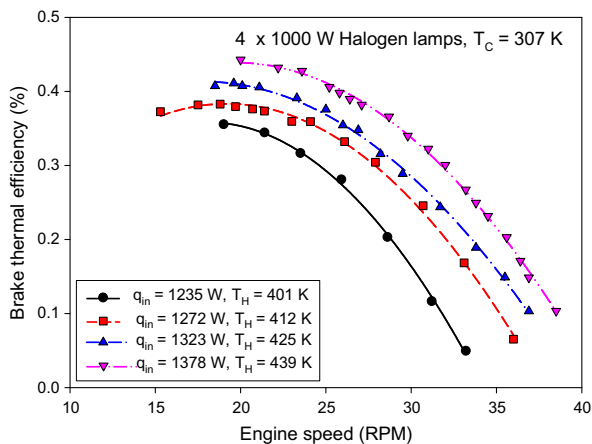


Fig. 11. Variations in brake thermal efficiency at various actual heat inputs.

maximum engine torque, shaft power, and brake thermal efficiency increases with increasing heater temperature.

The main technical problem is that the engine gives very low brake thermal efficiency (1.5% of Carnot efficiency, approximately). This may be caused by low shaft power

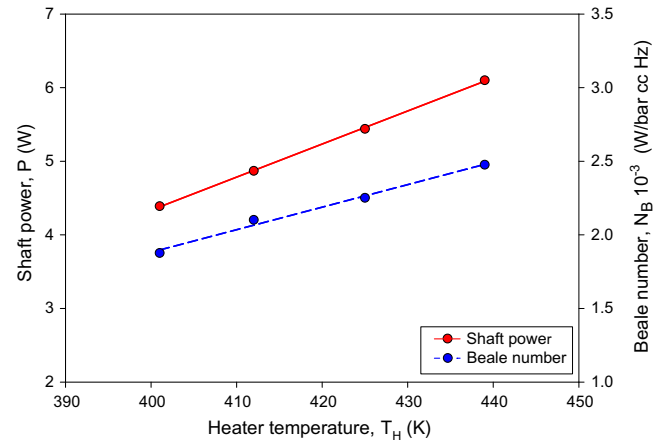


Fig. 12. Variations in engine maximum shaft power and Beale number with heater temperature.

due to high friction loss between the power pistons and cylinder. It is also very difficult to align four power pistons, which are rigidly connected as single members in two sets, to the four separately mounted cylinders. Another cause is that the engine operates at a relatively low-temperature. The heat source efficiency, the distance from lamp to displacer head, the displacer head thickness, and convection heat loss also affected the brake thermal efficiency.

Performance improvement in terms of design and construction can be achieved in many ways. For example, the alignment and precision of engine parts can be improved by using standard parts (e.g. using standard rod ends, instead of connecting the large and small ends of the rods which are made from ball bearings) and professional technicians who have specific experience in constructing or rebuilding engines. In addition, friction loss at a displacer guide rod can be reduced by changing the seal used at the displacer rod from a rubber seal to an oil grooves seal, as used in the seal of power-piston. Moreover, flywheel weight can be reduced by decreasing the weight of the power-piston, which can be done by making piston skirt and piston head thinner and by strengthening them with reinforced stiffeners. The displacer weight can also be reduced by changing the regenerator matrix from stainless steel to aluminum.

The four power-piston engine developed is specifically designed to have four power cylinders directly installed on a cooler plate coupled on a displacer cylinder. Thus, it is not necessary to use transfer ports, which results in as minimal dead volume as possible. This engine design is based on a principle of multifunctional capability of parts. Making a cooler plate part of a cooler not only helps in reducing the number of parts, but also helps in ventilating heat from the power cylinders. Furthermore, the displacer is also designed to serve as a regenerator. As a result, not only the engine structure is simple and uses as minimal parts as possible but the production cost is also lower.

The four power-piston configuration is good in that it yields as much power as a four-cylinder single-acting

engine or a two-cylinder double-acting engine. However, both the four-cylinder single-acting engine and the two-cylinder double-acting engine have to use four displacer cylinders. Hence, it is very difficult or even impossible to use those engines with a conventional solar concentrator.

The four power-piston engine developed is planned to be tested in the next step with a solar concentrator. Therefore, this engine is more likely to be developed into a compact engine with a high power-piston swept volume that can be used with a conventional solar concentrator. Since the LTD Stirling engine can work in low-temperature, it is possible to use a simple solar concentrator such as the conical reflector, which has a structure that is easier to construct than the parabolic dish concentrator. Therefore, the production cost of this part is also lower.

5. Conclusions

A kinematic, single-acting, four power-piston, gamma-configuration LTD Stirling engine was tested with a solar simulator using non-pressurized air as a working fluid. Four 1000 W halogen lamps were used in the solar simulator. The engine was tested with four different simulated solar intensities. Results from this study indicate that the engine performance and heater temperature increase with increasing simulated solar intensity. In fact, findings indicate that the maximum engine torque, shaft power, and brake thermal efficiency increases with increasing heater temperature. At the maximum simulated solar intensity of 7094 W/m², or actual heat input of 1378 W and a heater temperature of 439 K, the engine produces a maximum torque of 2.91 N m, a maximum shaft power of 6.1 W, and a maximum brake thermal efficiency of 0.44% at 20 rpm, approximately.

Although this engine performance is not so high, if we consider the fact that the solar-powered Stirling engine is powered by an emission free heat source, this study is a worthwhile step towards clean energy production. Furthermore, this engine design gives a compact LTD Stirling engine with high power-piston swept volume that could possibly be used with a simple conventional solar concentrator, the structure of which is easier to construct. An example of this is the conical reflector.

Besides increasing the precision of engine parts, the engine performance can be improved by increasing the heat source efficiency. By using a transparent cover for the bottom of the displacer head or absorber, for example, will enhance the heat transfer to the engine and potentially improve the engine performance. The engine performance could be further increased if a better working fluid, e.g. helium or hydrogen, is used instead of air and/or by operating the engine at varying degrees of pressurization.

Acknowledgements

The authors would like to express their appreciation to the Joint Graduate School of Energy and Environment (JGSEE) and the Thailand Research Fund (TRF) for providing financial support for this study.

Appendix A. Supplementary material

Supplementary data associated with this article can be found, in the online version, at [doi:10.1016/j.solener.2007.12.005](https://doi.org/10.1016/j.solener.2007.12.005).

References

- Haneman, D., 1975. Theory and principles of low-temperature hot air engines fuelled by solar energy. Report Prepared for US Atomic Energy Communication Contract W-7405-Eng-48.
- Iwamoto I., Toda F., Hirata K., Takeuchi M., Yamamoto T., 1997. Comparison of low- and high temperature differential Stirling engines. In: Proceedings of the 8th International Stirling Engine Conference, pp. 29–38.
- Kongtragool, B., Wongwises, S., 2003a. A review of solar-powered Stirling engines and low temperature differential Stirling engines. *Renewable and Sustainable Energy Reviews* 7, 131–154.
- Kongtragool, B., Wongwises, S., 2003b. Theoretical investigation on Beale number for low temperature differential Stirling engines. In: Proceedings of The 2nd International Conference on Heat Transfer, Fluid Mechanics, and Thermodynamics, Paper no. KB2, Victoria Falls, Zambia.
- Kongtragool, B., Wongwises, S., 2005a. Investigation on power output of the gamma-configuration low temperature differential Stirling engines. *Renewable Energy* 30, 465–476.
- Kongtragool, B., Wongwises, S., 2005b. Optimum absorber temperature of a once-reflecting full conical concentrator of a low temperature differential Stirling engine. *Renewable Energy* 30, 1671–1687.
- Kongtragool, B., Wongwises, S., 2007a. Performance of low temperature differential Stirling engines. *Renewable Energy* 32, 547–566.
- Kongtragool, B., Wongwises, S., 2007b. Performance of a twin power-piston low temperature differential Stirling engine powered by a solar simulator. *Solar Energy* 81, 884–895.
- O'Hare, L.R., 1984. Convection powered solar engine. US Patent, pp. 4, 453, 382.
- Rizzo, J.G., 1997. The Stirling Engine Manual. Camden miniature steam services, Somerset, pp. 43. 153–155.
- Senft, J.R., 1991. An ultra low temperature differential Stirling engine. In: Proceeding of the 5th International Stirling Engine Conference, Paper ISEC 91032, Dubrovnik, May.
- Senft, J.R., 1993. Ringbom Stirling Engines. Oxford University Press, New York, pp. 72, 88, 110, 113–137.
- Spencer, L.C., 1989. A comprehensive review of small solar-powered heat engines: Part III. Research since 1950-“unconventional” engines up to 100 kW. *Solar Energy* 43, 211–225.
- Van Arsdell, B.H., 2001. Stirling Engines. In: Zumerchik, J. (Ed.), *Macmillan Encyclopedia of Energy*, vol. 3. Macmillan Reference USA, pp. 1090–1095.
- White, E.W., 1983. Solar heat engines. US Patent, pp. 4, 414, 814.